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DESIGN AND ANALYSIS OF PRESSURE VESSEL

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ABSTRACT

The main purpose of the project is to design a pressure vessel according to ASME standards and do thermal analysis on the pressure vessel made up of different types of materials by varying the shell thickness of the vessel for various ambient temperature. Thus, comparing the results to find the optimum thickness for which the pressure vessel is safe to use in industry. A worst case scenario is also considered for which a crack is present on the external surface of the pressure vessel. Fracture mechanics is used to analyze the different geometry of crack for different kinds of material. Thus, predicting the extent to which the pressure vessel is safe for use.

KEYWORDS: Pressure Vessel, Design, Fabrication & ANSYS

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Original Article

INTRODUCTION

Pressure vessel is an enclosed unit in which the pressure acts from inside or outside the enclosed volume. They are most commonly used in industries as heat exchangers, reactors, storage vessels, etc. Due to pressure acting, there is a chance for the contained fluid to leak out which can cause serious accident and loss of life. For this reason, the design, fabrication and testing techniques are controlled by few legislation organizations like ASME, BS, and API standards etc. All the pressure vessels used in the industries must be certified by either one of the legislatures.

LITERATURE REVIEW

Devaraju and Pazhanivel (2015) have studied stress analysis on pressure vessels by considering the internal pressure, self-weight and the fluid weight. They have designed the pressure vessel using manual calculations and compared these computed stress values with the results obtained from the ANSYS software. They concluded that the stress acting on the shell of the pressure vessel designed is much less than that of the allowable stress of the material. Thus, the pressure vessel is safe for the usage. Nitinchandra et al. (2013) have investigated pressure vessel for marine substation applications considering different materials. The stress analysis has been performed considering internal and external pressure for these different materials by placing pressure vessels at different depths of the ocean from the sea surface using MATLAB software. The external pressure applied is caused due to the pressure caused by sea water at that depth and the internal pressure is caused due to gas induced inside the vessel in order to balance the external pressure. They have concluded that the pressure vessel designed will be helpful for marine applications.

Sadanandam et al. (2017) has done research on design and analysis of pressure vessel using finite element method. They have sub-divided the vessel into smaller elements and applied the internal pressure in order to

analyze the stress. They used principal stress theory and distortion energy theory for validating their design and the calculated results were compared with the results from the FEA software. They concluded that the maximum principal stress as per manual calculation was in line with the FEA results and hence the pressure vessel design considered was safe. Sandeep gond et al. (2004) has done analysis on the pressure vessel. The selection of the material, design and stress calculation of the pressure vessel was done as per the ASME standards. They have also intended to prove that the multilayer pressure vessels are capable of withstanding a higher internal pressure rather than that of the solid wall. They have done the analysis of the pressure vessel considering different materials in order to reduce the cost of construction. They concluded that the maximum stress developed in pressure vessel was within the yield stress of the material.

Sheik Abdul and Chandra Sekhar (2012) have studied the structural analysis due to change in location of the nozzle in a pressure vessel. They intend to find the location of the nozzle for which the stress value is minimal. The pressure vessel analyzed was filled with water with a working pressure of 9 Kg/cm^2 and internal diameter of 100 cm. They concluded that a minimum of 8 mm wall thickness is enough to hold the pressure vessel with a low value of factor of safety. Apurva et al. (2018) has done research on design and analysis of the pressure vessel. Their main focus is to analyze the safety parameters of the pressure vessel for a given working pressure. They have taken the main parameters that effect the safety of the pressure vessel like, material selection, design and fabrication. They have designed the pressure vessel using seamless pipe instead of making the shell using a plate. They concluded that the maximum working pressure considered was within the allowable limit. Merlin and chitaranjan (2017) have studied different types of end domes in the analysis of pressure vessel considering torispherical and hemispherical heads. They considered the pressure vessel to be non-linear symmetric and material to be nonlinear and focused on finding the optimum minimum thickness. They concluded that the stress accumulated in the torispherical head was minimum as compared to hemispherical head.

Rashmi and vinod (2017) have done analysis on pressure vessel in order to find the difference between flat head and hemispherical head. They have done the analysis considering different orientation and different number of saddle support. They concluded that the Von-Moses and normal stress of the pressure vessel are almost same for both flat head and hemispherical head whereas, the stress at the closure of flat head is found to be almost double on the hemispherical head vessel. Anandhu and avis (2017) have studied the analysis of horizontal oriented pressure vessel. The pressure vessel designed was modeled using CATIA and analyzed using ANSYS software. They concluded that the pressure vessel with shell thickness of 18 mm is safer as compared to 16 mm shell thickness, whereas, the shell thickness of 16 mm is much optimized to use while considering the economy of construction of pressure vessel.

Siva krishna and seshaiah (2012) have studied multilayer pressure vessel that can withstand a high pressure. They designed the pressure vessel using ASME standards for designing and checking various parameters of pressure vessel. The finite element analysis was used to analyze both the solid pressure vessel and multilayer pressure vessel. They concluded that the internal stress formed in multilayer pressure vessel was much lesser when compared to solid pressure vessel. Kuhn et al. (2000) has studied the design and analysis of full composite pressure vessel using FEA. They studied the different types of end domes for optimizing the weight and material variation. They concluded that the composite material can be used for optimizing the weight in case of pressure vessels. Wadkar et al. has studied design and analysis of pressure vessel using ANSYS software. They studied the stress concentration of a pressure vessel considering the shell and end dome by comparing the results of ANSYS with the manual calculations. They concluded that the stress concentration was very less and the pressure vessel is safe for use in industries.

METHODOLOGY

The analysis of the pressure vessel was carried out using ANSYS. The material considered for the analysis are ASME Grade material, Stainless Steel, Copper Alloy, Aluminium Alloy, Grey Cast Iron and Titanium Alloy. The material properties considered were having a yield stress of 260 MPa and poison's ratio considered was 0.3.

Design Calculations

The pressure vessel is designed to meets all the requirement of ASME standards. The minimum thickness of the shell required considering cylindrical shells with longitudinal joint was estimated using.

$$\mathrm{t} = \frac{\mathrm{PR}}{\mathrm{SE}-0.6\mathrm{P}} + \text{Corrosion Allowance} \tag{1}$$

Where t is the minimum thickness of the shell, P is the internal design pressure, R is the inner radius of the shell, S is the maximum allowable stress value of the material and E is the joint efficiency of the vessel.

The minimum thickness of the pressure vessel for external pressure was calculated using

$$\mathrm{P}_a = \frac{4 \times B}{3 \times \left(\frac{D_0}{\mathrm{t}}\right)} \tag{2}$$

Where P_a is the external pressure on the pressure vessel, B is the factor for maximum design metal temperature, D_o is the outside diameter of the pressure vessel and t is the thickness of the shell for external pressure.

The minimum thickness for the dished end for internal pressure is calculated using the following equation

$$t = \frac{PD}{2SE-0.2P} + corrosion\ allowance \tag{3}$$

Where t is the minimum thickness required for dished end, P is the internal design pressure, D is the inner diameter of the shell, S is the maximum allowable stress value of the material and E is the joint efficiency.

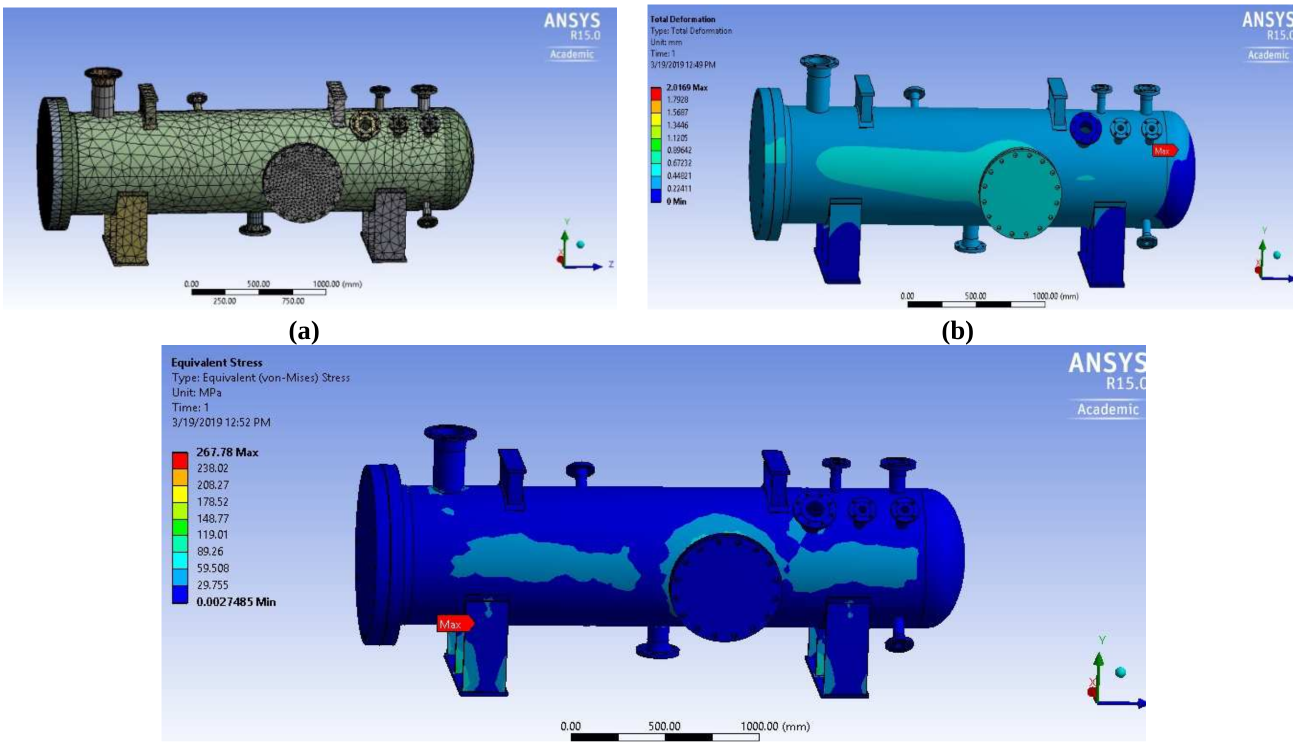
The minimum thickness of the dished end due to external pressure is calculated using

$$\mathrm{P_a} = \frac{\mathrm{B}}{\left(\frac{\mathrm{R_0}}{\mathrm{t}}\right)} \tag{4}$$

Where t is the minimum thickness for dished, R_o is the equivalent radius of the dished end (R_o = k * diameter, where k is factor depending on ellipsoidal head, value obtained from Table: UG-33.1)

Thermal Stress Analysis of Pressure Vessel

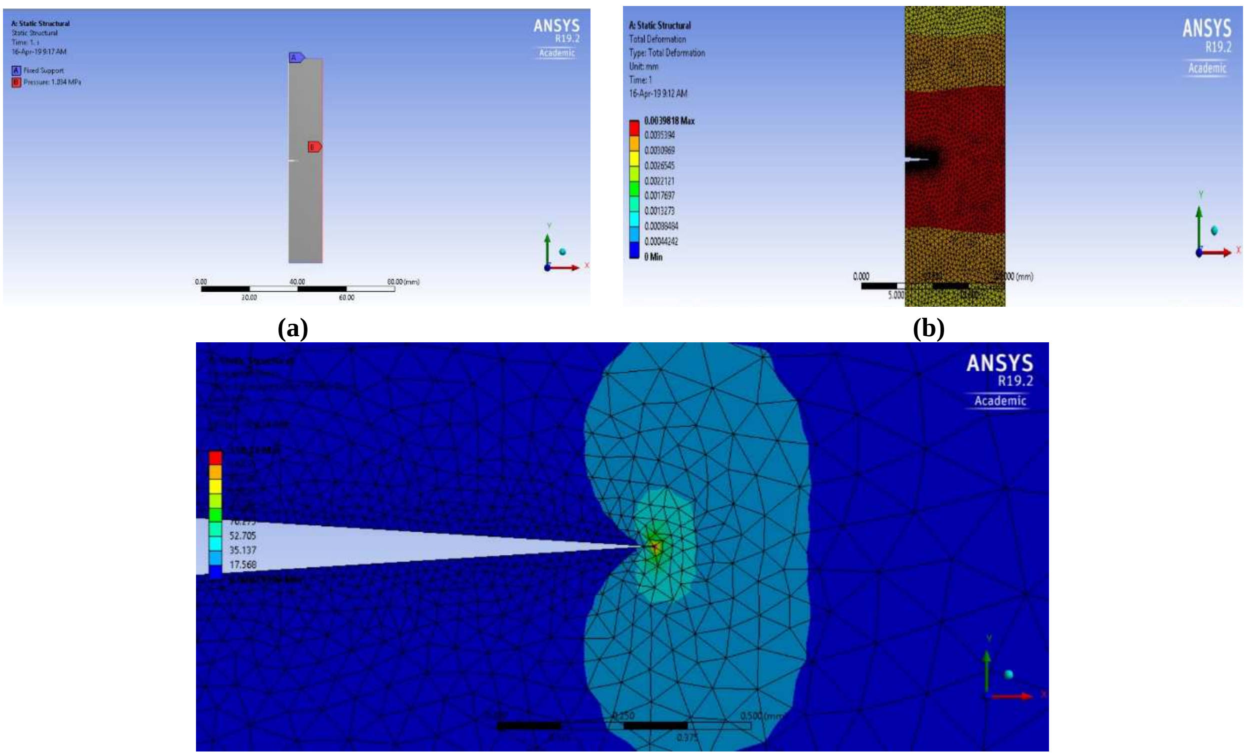
Thermal Stress analysis has been carried out for Pressure Vessel with different shell thickness. As per the design calculation, the minimum thickness required for the pressure vessel is found to be 10.92 mm which includes corrosion allowance. Thus, we consider the thickness starting from 8 mm till 25 mm for the analysis purpose. We also consider for different ambient temperatures of 0 ^{0}C, 25 ^{0}C and 50 ^{0}C. Convection is applied on the internal surface of the pressure vessel using stagnant air of 150 ^{0}C. The saddles of the pressure vessel are fixed support and pressure of 1.034 MPa is applied on internal surface of the pressure vessel. Thermal Stress Analysis is performed to obtain deformation and Thermal Stress formed on Pressure Vessel.



(a) (b)

(c)

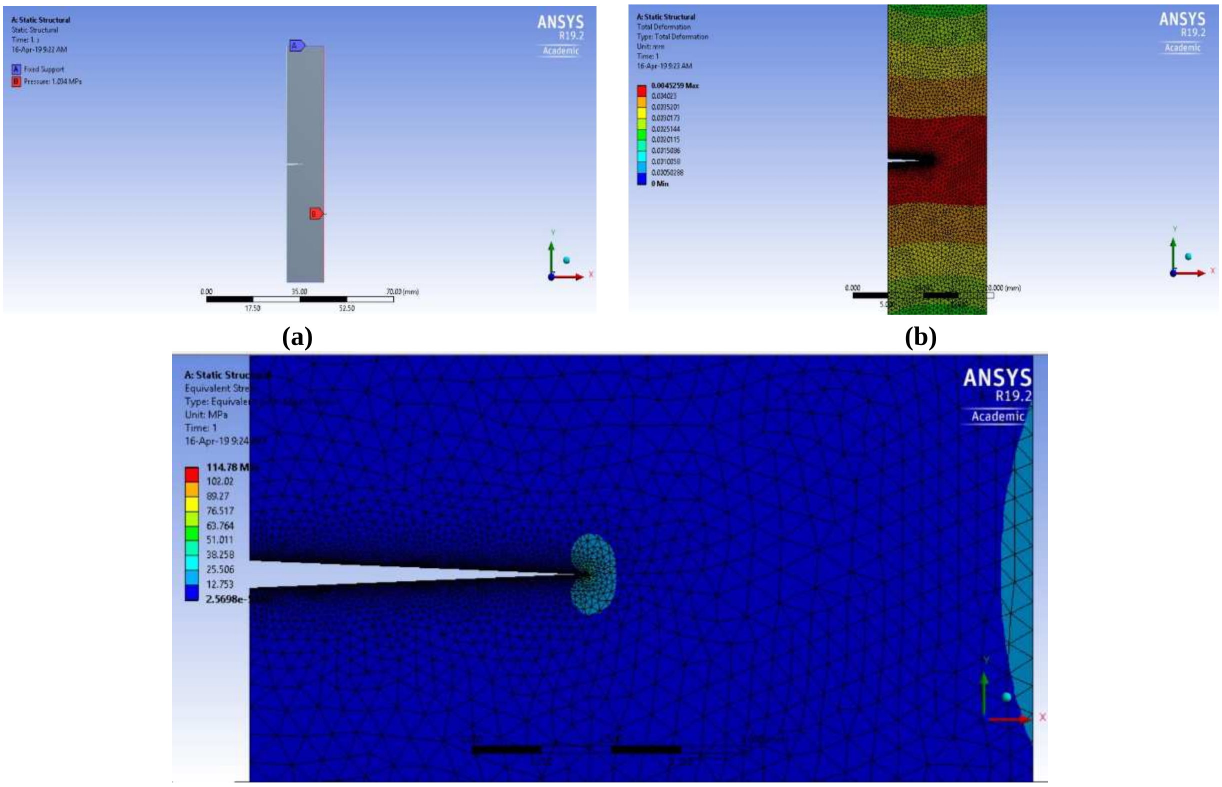
Figure 1: Thermal Stress Analysis of ASME Material for 14 mm Thickness (a) Mesh with Element size of 100 mm (b) Total Deformation (c) Thermal Stress Analysis.



(a) (b)

(c)

Figure 2: Crack Propagation Growth Considering 4 mm (a) Plate with Fixed Support and Pressure (b) Total Deformation (c) Static Stress Analysis.



(a) **(b)**

(c)

Figure 3: Crack Propagation Growth Considering 7 mm (a) Plate with Fixed Support and Pressure (b) Total Deformation (c) Static Stress Analysis.

Crack Propagation in Pressure Vessel

The crack propagation was analyzed along the thickness of the specimen. The thickness of the pressure vessel considered was 14 mm. Along with this the, thickness of the specimen crack with a depth of 4 mm and 7 mm have been considered for fracture analysis. Static structural analysis with pre meshed fracture mechanics have been performed to analyze the crack growth for the above crack depth. A pressure of 1.034 MPa is applied on the internal surface of the thickness of the specimen and in turn static structural analysis was performed. The corresponding total deformation, Equivalent stress and stress intensity factors have been analyzed for different depths and widths of crack.

From the Figure 2 (c), we can note that the stress distribution formed on tip of the crack. It is noted that the maximum stress is formed at the tip of the crack and when this stress acts for a long time, due to residue stress on the tip of crack, the region becomes plastic in nature. Thus, after this stage, as the stress starts to act on the plastic region formed, the crack propagation takes place. Due to this, the depth of the crack keeps increasing with time along the formation of the plastic region.

From the Figure 3 (c), it is noted that the stress distribution has been reduced with increase in depth of crack. This is because of formation of plastic region in where less stress is formed.

RESULTS AND DISCUSSIONS

Thermal Stress Analysis of pressure vessel for different types of material by varying the shell thickness of the pressure vessel for the real life boundary conditions has been carried out. The below Table 1, gives us the values of deformation and thermal stress formed on the pressure vessel. As we reach the thickness of 25 mm, it is noted that for ASME material,

the stress becomes very low, which indicates that, the pressure vessel can be used for a long period of time. If we further increase the thickness of the shell, the changes in thermal stress will be negligible. Thus we conclude Thermal Analysis of the pressure vessel by 25 mm. The different temperatures are considered as the pressure vessel can be used in any part of region from the lowest to highest temperature regions in the world. In case, the pressure vessel is used in colder region of below 0 Deg C, the material should be impact tested for the safety of the pressure vessel.

Table 1: Thermal Stress Analysis Results

Material	Thickness (mm)	0 Deg C		25 Deg C		50 Deg C	
		Deformation (mm)	Thermal Stress (Mpa)	Deformation (mm)	Thermal Stress (Mpa)	Deformation (mm)	Thermal Stress (Mpa)
ASME Material	8	2.10	181.42	1.47	145.90	0.91	129.39
	10	1.94	172.47	1.24	137.41	0.81	114.65
	12	1.79	157.01	1.17	129.33	0.75	101.90
	14	1.64	102.40	1.10	98.31	0.69	92.45
	20	0.42	78.23	0.40	70.27	0.32	62.33
	25	0.38	57.47	0.36	52.44	0.27	46.59
Stainless Steel	8	5.09	1359.90	4.29	1135.60	3.46	911.34
	10	5.06	1340.70	4.27	1116.30	3.49	891.83
	12	5.06	1252.80	4.25	1046.10	3.45	839.35
	14	5.03	1145.70	4.25	956.60	3.45	767.36
	20	4.12	987.42	3.87	814.49	3.01	689.40
	25	3.05	823.67	2.97	694.82	2.12	569.42
Aluminium	8	7.29	695.09	6.23	581.69	5.18	468.30
	10	7.22	674.49	6.15	561.00	5.09	447.51
	12	7.19	635.24	6.11	531.49	5.05	427.74
	14	7.16	575.39	6.08	481.29	5.01	387.18
	20	6.48	442.89	4.98	386.11	3.74	293.47
	25	5.83	328.38	4.31	273.47	2.88	173.44
Copper	8	5.59	846.69	4.75	708.06	3.92	569.24
	10	5.54	825.73	4.70	686.99	3.86	548.26
	12	5.52	779.19	4.68	649.58	3.84	519.96
	14	5.51	711.69	4.66	591.10	3.82	470.68
	20	4.89	603.40	4.08	453.44	3.38	378.94
	25	4.43	488.97	3.74	378.31	2.97	302.56
Gray Cast Iron	8	3.67	498.78	3.16	417.89	2.71	337.00
	10	3.61	480.40	3.11	399.45	2.61	318.51
	12	3.58	463.67	3.07	388.39	2.57	316.48
	14	3.55	428.56	3.04	358.89	2.54	295.96
	20	3.17	368.31	2.64	291.64	1.83	240.87
	25	2.78	309.82	2.32	238.77	1.47	183.34
Titanium	8	3.28	401.02	2.95	336.77	2.68	272.53
	10	3.22	378.69	2.80	314.37	2.47	250.06
	12	3.19	366.15	2.76	305.40	2.34	252.53
	14	3.16	331.28	2.73	274.03	2.21	232.53
	20	2.82	285.45	2.32	238.75	1.98	186.34
	25	2.47	224.78	2.04	174.34	1.74	141.13

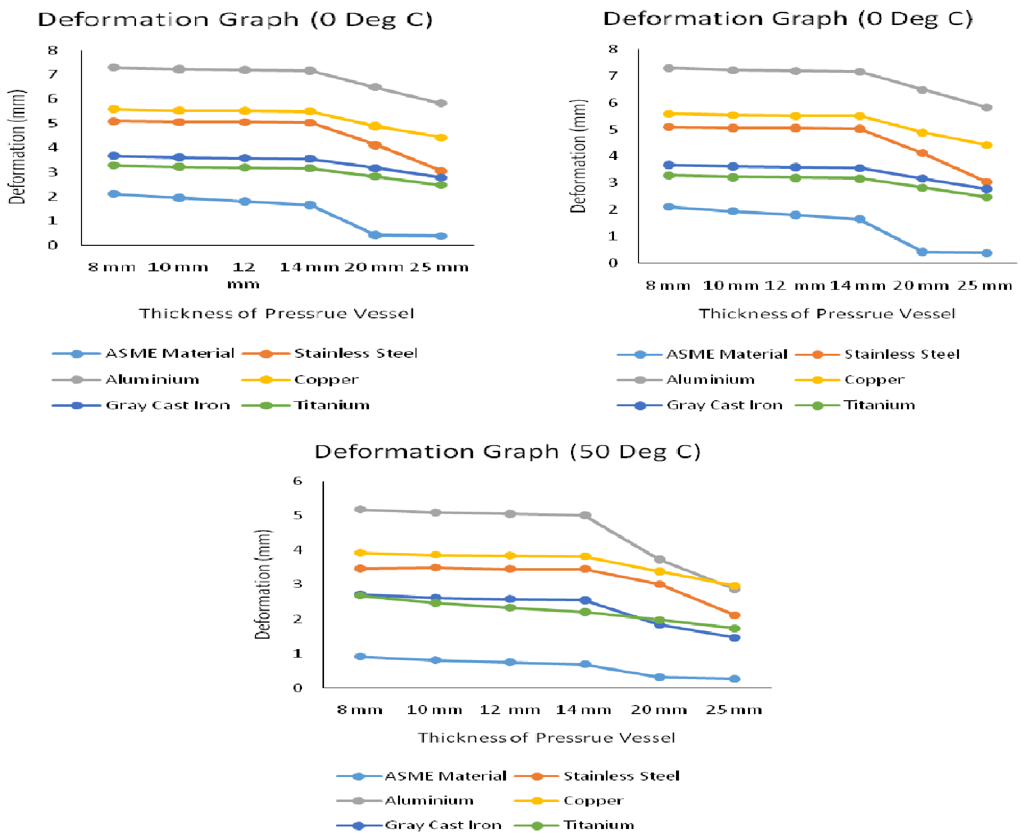


Figure 4: Deformation Graphs of Different Materials at Different Ambient Temperatures.

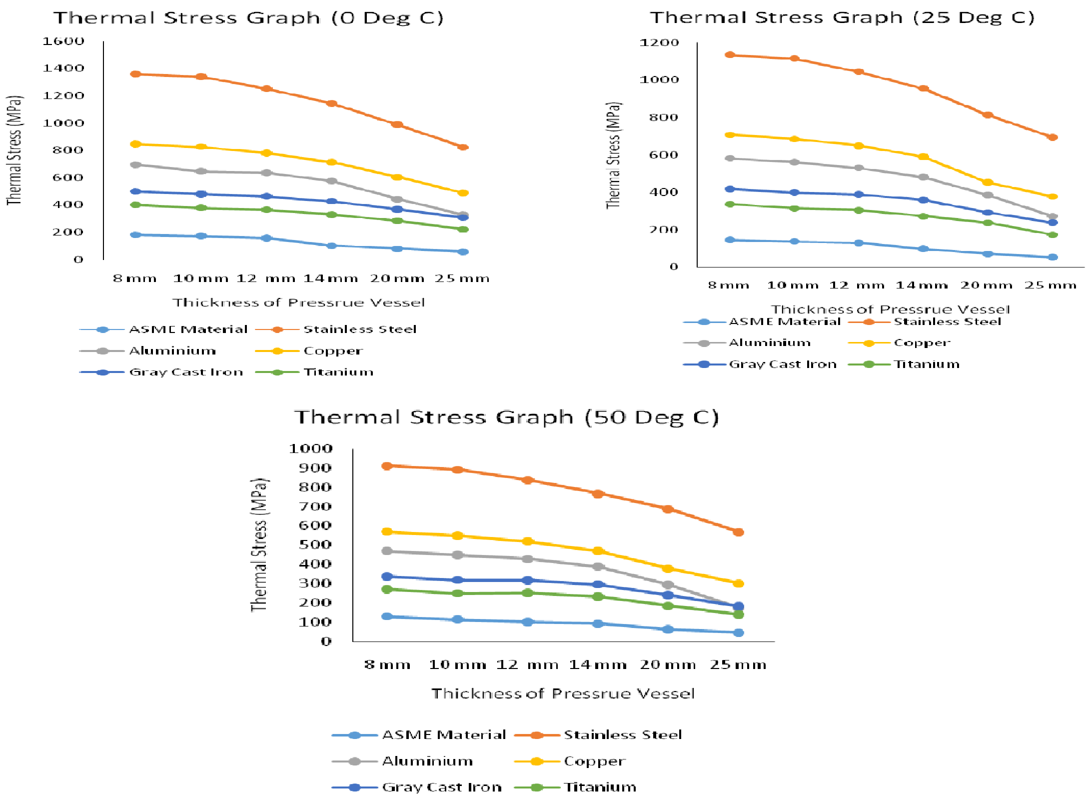


Figure 5: Thermal Stress Graphs of Different Materials at Different Ambient Temperatures.

From the deformation graphs Figure 4, we can see that, Aluminium has the maximum value of deformation. The deformation graphs for 3 different ambient temperature have similar curves, whereas, there is difference in the deformation values for different temperature. This is due to the thermal expansion property of the material used. It is found that, the lower 3 lines in the graphs, which represent ASME material, Titanium Alloy and the gray cast iron are more reliable for use in the industry. Whereas a pressure vessel made using titanium alloy is much costlier and by considering the thermal stress into account, the thermal stress in case of the titanium alloy is much higher. Thus, the pressure vessel manufactured using the titanium alloy is less reliable.

It can be seen from the thermal stress graph Figure 5, the thermal stress formed in pressure vessel fabricated using the stainless steel material is having a huge stress value, which is unacceptable. This thermal stress is formed due to the thermal expansion of the pressure vessel, which in turn causes as higher stress accumulation in the saddle of the pressure vessel. The stainless steel material is majorly used for fabricating the vertically oriented pressure vessels. ASME SEC II provides different types of stainless steel materials, which are based upon the ASME rules and regulations. The pressure vessel manufactured using stainless steel should comply will all these ASME codes. It is also noted that, ASME has much lower Thermal stress of below 200 MPa, which makes it most reliable.

Fracture Analysis Results

Table 2: Comparison of Crack Tip Deformation with Different Types of Materials

Depth of Crack (mm)	ASME Material	Stainless Steel	Copper	Aluminium	Gray Cast Iron	Titanium
4.00	0.004	0.0041	0.0073	0.0112	0.0072	0.0083
4.25	0.004	0.0042	0.0074	0.0114	0.0073	0.0085
4.50	0.0041	0.0043	0.0075	0.0115	0.0074	0.0086
4.75	0.0042	0.0043	0.0076	0.0118	0.0076	0.0087
5.00	0.0043	0.0044	0.0078	0.012	0.0077	0.0089
5.25	0.0043	0.0045	0.0078	0.0121	0.0078	0.009
5.50	0.0044	0.0045	0.008	0.0123	0.0079	0.0091
5.75	0.0044	0.0046	0.0081	0.0125	0.008	0.0093
6.00	0.0045	0.0047	0.0082	0.0127	0.0082	0.0094
6.25	0.0046	0.0047	0.0083	0.0129	0.0083	0.0095
6.50	0.0046	0.0048	0.0084	0.013	0.0084	0.0097
6.75	0.0047	0.0049	0.0085	0.0132	0.0085	0.0098
7.00	0.0047	0.0049	0.0087	0.0134	0.0086	0.0099

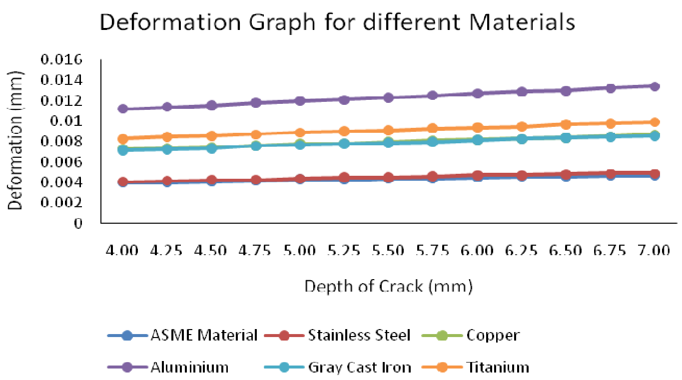


Figure 6: Comparison of Crack Tip Deformation with Different Types of Materials.

Table 3: Comparison of Stress Intensity Factor with Different Types of Materials

Depth of Crack (mm)	ASME Material	Stainless Steel	Copper	Aluminium	Gray Cast Iron	Titanium
4.00	26.19	26.177	26.137	26.15	26.216	26.108
4.25	26.48	26.472	26.43	26.443	26.514	26.398
4.50	26.72	26.708	26.661	26.677	26.753	26.629
4.75	26.88	26.869	26.819	26.836	26.916	26.784
5.00	26.98	26.964	26.911	26.929	27.015	26.874
5.25	26.96	26.94	26.884	26.903	26.995	26.844
5.50	26.87	26.85	26.79	26.81	26.908	26.749
5.75	26.69	26.672	26.608	26.63	26.733	26.565
6.00	26.44	26.419	26.352	26.374	26.483	26.306
6.25	26.06	26.038	25.968	25.992	26.106	25.92
6.50	25.6	25.581	25.508	25.533	25.652	25.457
6.75	25.05	25.026	24.949	24.975	25.1	24.896
7.00	24.41	24.385	24.305	24.332	24.462	24.25

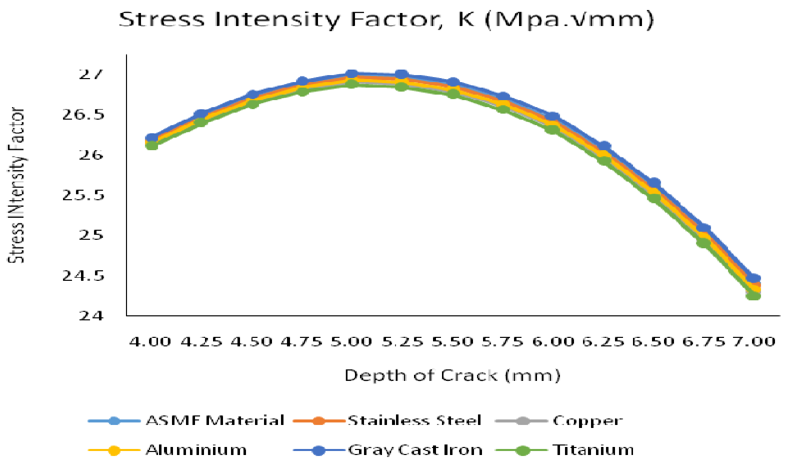


Figure 7: Stress Intensity Factor Graph for Different Materials.

From the Figure 6, it can be noticed that, Aluminium is having highest deformation values for the crack tip. As it is known that Aluminum has the highest elasticity properties as compared to the other materials, the deformation is maximum. It is also noted that, grey cast iron and titanium alloys have an intermediate values for deformation at the crack tip. Finally, the stainless steel and ASME material have very less deformation values as compared to other materials.

From the Figure 7, it can be seen that all the materials are having a similar curve of stress intensity factor. We can also note that for all the materials, the maximum value of the stress intensity factor is achieved at 5 mm of crack depth. We can assume that, the pressure vessel with a crack is safe for use till the depth of crack is about 5 mm, regardless of the width of the crack. While concerned about the safety of the pressure vessel with crack, the stress values should be also considered, which decides the rate of propagation of the crack. The crack should be continuously be monitored using Ultrasonic testing to monitor the depth of crack. For the safety reasons, we assume that pressure vessel should be reparied as the depth of crack reaches 4 mm, as failure of pressure vessel.

CONCLUSIONS

From the present project work, pressure vessels were designed as per ASME standards and according to the analysis done, the following conclusions have been made:

- It was found that from the design calculation, the minimum thickness required for the shell and dished end are about 10.99 mm including the corrosion allowance of 3 mm.
- It was found that the pattern of the graphs for the different materials are similar for the different ambient temperatures. This is due to difference in property of thermal expansion for different types of materials.
- The fracture analysis is done for different types of material using 14 mm thickness specimen. We found that the maximum crack tip deformation is found in case of Aluminium due to its high elastic properties and the lowest in the case of ASME material.
- It is found that the stress intensity factor increases initially with the increase in the depth of crack and the reaches the maximum values at a depth, which is considered to be the point at which the capability of the material to withstand the crack tip stress is maximum.
- For all the materials, the curve reaches the maximum point for same depth of crack for which the resistance for the crack propagation is maximum after which the propagation is rapid.

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